REVIEW OF CENTRIFUGAL COMPRESSORS HIGH PRESSURE TESTING FOR OFFSHORE APPLICATIONS

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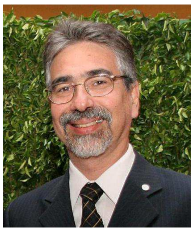

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ABSTRACT

There are many doubts about the risk / reward relation when it comes to ordering tests involving high pressure centrifugal compression systems.

There are different tests that can be classified in this category, namely Full Load, Full Pressure, Full Density, Full Speed or a combination of those along with the ASME PTC 10 Type 1 tests. Normally these tests are carried out as a Complete Unit Test (String Test).

This paper aims to discuss the advantages and disadvantages of the various types of tests.

The codes from API and ASME do not detail the requirements of full pressure tests. The installation, the procedure, and the acceptance criteria are subject of agreement between vendor and purchaser. Therefore the importance of a meticulous test description during the bid phase is also discussed.

In this paper, the various types of full load tests will be presented and along with a discussion on their capabilities to detect a variety of problems. Some issues which occurred during full load tests and at site will be described. Based on test characteristics and at the related cases, recommendations are made when ordering full pressure tests along with considerations regarding the return of investment of these tests.

INTRODUCTION

It is known that the ASME PTC 10 Type 2 performance test and the API 617 mechanical running test might not identify a series of problems. Regardless, there are still many doubts about the risks versus rewards when it comes to ordering tests involving high pressure centrifugal compressors. There are different tests whose main objective is to emulate the field conditions.

The ASME PTC 10 Type 1 test requires that the specified gas at or very near the specified operating conditions is used. The values for inlet pressure, inlet temperature, molecular weight, etc. have a constricted permissible deviation and the combination of these values shall not exceed the dimensionless parameters that will guarantee full similarity between test and site conditions. The efforts required to perform this kind of test are enormous. For some cases, the use of hydrocarbon gas at the OEM test facilities is problematic and requires many measures to address HSE and / or local regulations. Maintaining the right gas composition is also not easy, because the loop test will leak (at the very least through shaft end seals) and may need to be replenished with gas during the test.

Other simplified high pressure tests do not have the strict requirements imposed by the ASME PTC 10 Type 1. These tests might be performed under full load, full pressure, full density, full speed or a combination of them. These tests can also be performed as a complete unit test in order to evaluate the entire train. The advantages and disadvantages of this type of tests will be further discussed.

The difficulties involved in preparing and performing an ASME PTC Type 1 test or a full load test will lead to higher costs and the introduction of such complex additional tests impacts directly on project schedule

Some will argue that the costs involved are not worthwhile. Given the difficulties of performing such tests, all site conditions, and therefore issues that may occur on normal operation, will not be covered and investigated. Their expressed opinion is that the normal test routine (ASME PTC 10 Type 2 performance test and the API 617 mechanical running test) is enough to analyze the overall condition of the compressor, especially when the compressor is not designed to achieve high pressure levels at site. However, if the normal test routine fails to identify problems with the equipment, they will be forced to resolve the issues in the field, with greater effort and costs.

TYPES OF TEST

ASME PTC 10 TYPE 1 TEST

As already briefly discussed, the ASME PTC 10 Type 1 tests are performed with the specified gas at, or very near, the field operating conditions. The permissible deviation for inlet pressure, inlet temperature, speed, molecular weight and capacity are shown on PTC10 table 3.1. In addition, the combined effect of inlet pressure, temperature and molecular weight shall not exceed 8% of deviation in the inlet gas density.

Table 1- ASME PTC 10 Type 1 (table 3.1) tolerances

Variable	Symbol	Units	Permissible Deviation
Inlet pressure	p_i	psia	5%
Inlet temperature	T_i	°R	8%
Speed	N	rpm	2%
Molecular Weight	MW	lbm/lbmole	2%
Cooling temperature difference		°R	5%
Coolant flow rate		gal/min	3%
Capacity	q_i	ft³/min	4%

In addition to all the parameters of ASME PTC 10's table 3.1, the combination of these values shall not exceed the parameters of code's table 3.2, that shows limits for specific volume ratio, flow coefficient, machine Mach number and machine Reynolds number. These parameters guarantee that the test is in similitude conditions with site and the performance can be confidently evaluated.

The test evaluates the compressor performance from overflow (or overload) to close to the surge condition. The mechanical integrity of components is also evaluated, since the compressor is operating at full pressure and full speed. The rotordynamic behavior can also be verified, although not the stability margin.

Full-pressure / Full-load / Full-speed Test (FPFLFS)

API 617 refers to a Full-pressure / Full-load / Full-speed test (chapter 1, item 4.3.8.6). The standard states that the conditions for this test should be discussed and developed jointly by the purchaser and the vendor. The discussion below addresses each one of these conditions, some additional tests and how the conditions can be combined to develop a test that meets the purchaser objectives while taking into account the OEM's test facilities limitations, as well as limiting cost. Since the combination of these requirements can result in a very different type of test, in this work we are going to refer to these tests in a generic way as "Full Load test"

Full Load (FL)

The term full load refers to an absorbed power equal to or higher than specified. This requirement alone is very vague as it does not give an indication on which parameters will be evaluated by the test. A vibration and mechanical assessment might be possible but, as mentioned, the term alone is not precise in defining the test procedure and what to expect from it. Therefore, it should be combined with other requirements to fulfill the purchaser objectives.

Full Speed (FS)

This requirement is especially important regarding the vibration evaluation. Although most machines that undergo a FPFLFS test would already have passed a mechanical running test at maximum continuous speed, such tests could replace the mechanical running test without compromises.

In some cases, it might be necessary to reduce the speed during certain steps of the test. As an example, if the test procedure requires an investigation on the low flow portion of the operating map, the speed might have to be reduced to avoid over-pressurizing the test loop. In most cases, the test is conducted on an inert gas medium having a higher *k* value than the specified gas resulting in higher discharge temperature. Discharge temperature might also limit the test speed.

Full Pressure (FP)

The full pressure condition permits the evaluation of the compressor for the mechanical integrity of its components, such as pressure containing components (not only the casing), dry gas seals, as well as bearings mechanical behavior and temperature rise. This is an important verification, especially for the axial bearing, which is not subjected to significant load during the mechanical running test or the performance test.

A full pressure test would also improve the evaluation of the rotordynamic behavior, since substantial aerodynamic cross-coupling forces, typically absent in a MRT, are introduced to the system.

Full Density (FD)

Although the API 617 8th Edition (2014) does not specify the term "full density", it does use the position of the compressor on a plot of flexibility ratio (the ratio of the maximum continuous speed divided by the first critical speed on rigid supports) vs. average gas density in the Level I screening criteria. API 617 8th Edition says to use the machine condition at the normal operating point unless the supplier and purchaser agree upon another operating point. The position of the compressor on this plot does not determine if the compressor will be stable or not. The position on the plot and the results of an API 617 Level I log dec analysis with the journal bearings, squeeze-film dampers (if used) and oil-film seals (if used) and an anticipated cross-coupling as in Memmott (2000), API 617 7th Edition (2002), and API 617 8th Edition (2014) are used to determine if a more detailed API 617 Level II log dec analysis with inclusion of the gas annular seals is needed.

Rather than straight line curves, for "Typical" and "Worst Case" lines as in the paper by Fulton (1994), the API plot has regions A and B, and an average gas density above which the compressor is always in Region B no matter what the flexibility ratio. Stricter analytical criterion is applied in Region B if one wants to avoid a Level II analysis.

The papers by Memmott (2002, 2010, and 2011) show a large amount of experience on the API plot. A case history is given in the 2002 paper of a high density CO_2 compressor that was in Region B and although below Fulton's worst case acceptable line needed the field application of stability enhancing features, squeeze-film dampers and shunt holes. The shunt holes were to the toothed labyrinth balance piston seal of this in-line compressor. The compressor had toothed labyrinth casing end seals.

The mechanical behavior of the compressor at site conditions may not be represented when only a discharge pressure criterion is used, since depending on gas composition and temperature the density may be lower than the one expected at site.

Figure 1 shows a sample of the compressors made by the last two author's company, on the API plot as in Memmott (2011). The compressors on the plot all have dry gas seals or toothed labyrinths as the casing end seals. Most all of them are with dry gas seals. The average gas density was not always calculated from the rated point; it may be for the highest anticipated cross-coupling or for the highest average gas density. Also added are more recent compressors, from the papers by Colby et al (2012) and Noronha et al (2014). The ones in circles have damper bearings, the ones in squares have non-damper bearings, and if hollow they have hole pattern or honeycomb seals at the division wall or balance piston. There are many more applications with hole pattern seals than honeycomb seals. All of them have dry gas or toothed labyrinth casing end seals, mostly all with dry gas seals. Shunt holes and swirl brakes are routinely applied at the division wall or balance piston.

The one on the right side of the plot is believed to be the highest average gas density of any centrifugal compressor application and was tested up to an average gas density of 530 kg/m^3 (33.06 lb/ft^3). It is discussed in the paper by Colby et al (2012). The paper by Memmott (2011) discusses some of the ones for which magnetic bearing exciter tests were done in FLFPFS shop tests to extract the log dec. For the three compressors on the right side of the plot this was done, but they are not in the 2011 paper. The 2011 paper also discusses other compressors for which there were FLFPFS tests but there were no magnetic bearing exciter tests The use of the hole pattern seals has extended the range of experience into the very high density region.

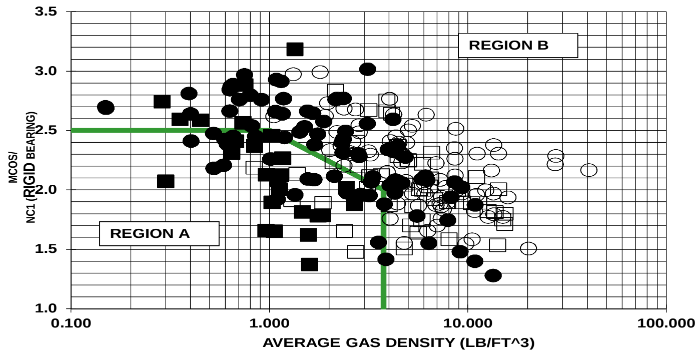

Figure 1 - API experience plot as in Memmott (2011).

Other OEM's have published their experience on the API plot, Camatti et al (2003), Moore et al (2006), Bidaut et al (2009), and Bidaut and Baumann (2010).

However, bearing span/average shaft diameter under the impellers vs. average gas density provides a more reliable guide to potential stability issues than flexibility ratio vs. average gas density as is shown in the papers by Memmott (2002, 2010 and 2011). See Figure 2 as in those papers with the same compressors as in Figure 1. Even if a compressor is not running fast compared to its first critical speed its shaft can still be very flexible and then the compressor needs close scrutiny.

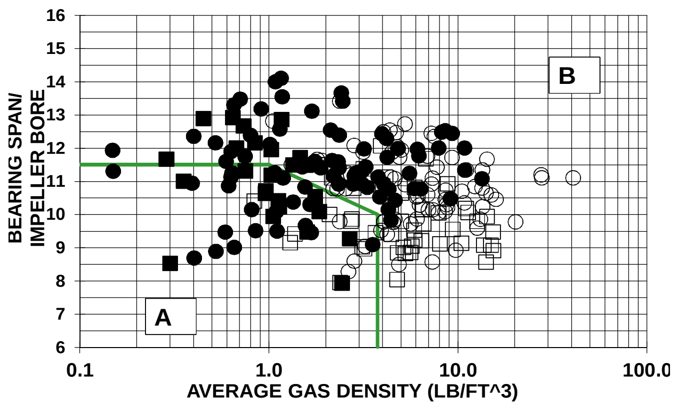

Figure 2 - Experience plot as in Memmott (2011) with bearing span/average shaft diameter under the impellers vs. average gas density.

Miranda and Noronha (2007) proposed that, given the difficulty to perform an ASME PTC 10 Type 1 test, a full density test could be used to verify the mechanical behavior and rotordynamic stability. In this work, limits for density were specified while other limits from the PTC Type 1 test were relaxed.

Complete Unit Test (CUT)

API 617 specifies, as an optional test, the complete unit test. This test is performed with the compressor, the transmission equipment, the main driver (electrical motor, gas turbine and so on) and any auxiliaries. This test is commonly denominated as a String Test by vendors and purchasers.

Some major components cannot be used during the test because of the test loop configuration, such as the anti-surge system and parts of the sealing system. Other subsystems need to be adapted in order to make the test possible, such as the software of the unit control panel (emulated signals, by-passes, etc.). This is one of the arguments offered by those who think that this test has more cost than benefits.

However, performing a CUT may be the only feasible way to conduct a high pressure test, especially for high power compressors, since adequate drivers are not always available at the vendors´ facilities.

In this test the rotor vibrations can be evaluated at near site conditions (electric motors and gearboxes are typically tested unloaded). The train can also be verified regarding correct calibration and installation of instruments, confirmation of dimensioning and installation of orifice flows, overall ergonomic arrangement, correct functioning of alarms, switchovers of pumps, filters, and coolers.

Concerns about torsional vibrations will require the complete train to perform any verification.

Stability Test (ST)

A stability test consists of the measurement of the logarithmic decrement (*log dec)* by using nonsynchronous forced excitation applied during the test. Stability tests can be done in conjunction with the traditional full load tests to measure the log dec, while the answer of a full load test when the stability test is not done is either stable or not stable (at the one specific condition). A main advantage of the stability test is to foresee the risk of instability in operational conditions other than the rated (7th Edition term) or normal operating point (8th Edition term). If stability testing was done only during a mechanical run test it would help to validate the combined rotor-bearing-pedestal system model, but it would not validate the dynamic modeling of the annular gas seals and that of the load related anticipated cross-couplings. The effect of the annular gas seals is critical for high-pressure high-density compressors.

Baumann (1999) presented results for log dec measurements on a compressor using an electromagnetic exciter. This compressor did not have a damper seal (honeycomb or hole pattern). Before these tests, log dec data was only available from laboratory test rigs.

Since 2001 magnetic bearing exciters have been used by the last two author's company in full load full pressure tests to validate the predicted log decs and the design of high pressure high density compressors for stable operation. All of those tests were conducted on compressors with hole pattern seals at the division wall or balance piston. At these places there were both shunt holes and swirl brakes. Some of the compressors had squeeze-film dampers in series with the journal bearings and all had dry gas casing end seals.

Data was taken at low, intermediate and full pressures. With the hole pattern seals, as the pressure and thus the density increased the log dec increased significantly, unlike what would be expected with toothed labyrinth seals. There was good agreement between the tested log decs, which were obtained with a frequency sweep and a single degree of freedom model for the data collection and the analytical log decs, which were obtained from the lateral rotor dynamic models of the rotor, journal bearings, squeeze-film dampers (if used) and annular gas seals. See the papers by Moore et al (2002), Moore and Soulas (2003), Gupta et al (2007), Soulas, et al (2011), Memmott (2011), Gupta (2011), and Colby et al (2012). Noronha, et al (2014) presented a paper on multi degree freedom modal testing with a magnetic bearing exciter of very high density compressors.

Another OEM has published results on stability testing (log dec measurements with magnetic bearing exciters) during FLFPFS tests with hole pattern seals. See the papers by Bidaut et al (2009), and Bidaut and Baumann (2010).

Pettinato, et al. (2010) presented results for an equivalent test by still another OEM, but in this case, the test was part of the shop order and the purchaser defined the acceptance criteria. This was a low pressure compressor.

Test Combinations

Typically, the above described tests are performed jointly. Common combinations for high pressure tests will be described and their characteristics will be evaluated in view of their capacity to diagnose potential operational problems. They will be presented from the simpler to the most complete test.

Full Load / Full Pressure / Full Speed

The parameters controlled during the test are power consumption, pressure and speed. The criterion normally applied is a zero negative tolerance for all parameters. The criterion for the suction pressure could be modified in view of another empirical stability criterion. Kirk and Donald (1983) defined a criterion, where they plotted a pressure parameter (the discharge pressure multiplied by the differential pressure across the compressor) vs. the flexibility ratio to evaluate stability.

The papers by Memmott (2002, 2010, 2011) present a different criterion with the same axes as the Kirk-Donald plot, which instead of Unacceptable and Acceptable lines have Regions A and B similar to those in the API 617 7th Edition plot, with the same criterion for going to a Level II from a Level I analysis as in API.

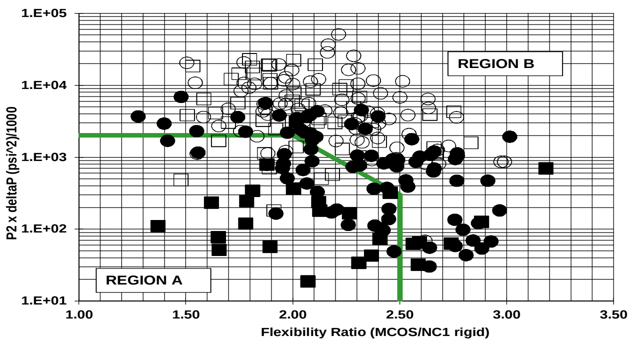

Figure 3 - Memmott's adaptation of the Kirk-Donald plot as in Memmott (2011)

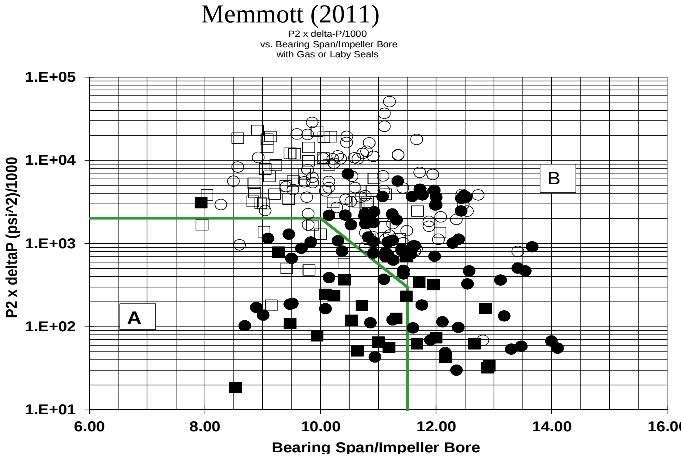

Figure 4 - Experience plot as in Memmott (2011) with pressure parameter vs. bearing span/average shaft diameter under the impellers

Figures 3 and 4 are as in Memmott (2011) with the same compressors as were shown in Figures 1 and 2. Figure 3 is the Memmott adaptation of the Kirk-Donald plot with pressure parameter plotted vs. flexibility ratio. Figure 4 shows experience on a plot of pressure parameter vs. bearing span/average shaft diameter under the impellers.

The simplest procedure for a full load / full pressure / full speed test is to operate the compressor during a period of four hours at only one point. This point normally corresponds to the specified pressure ratio or discharge pressure at site conditions. With those conditions satisfied, the test point is rarely close to the guarantee point. With this kind of test it is possible to have a partial evaluation of the mechanical behavior of the compressor, since some different issues may arise when moving the operating point in the compressor map.

When the compressor is subjected to different operating conditions, it is possible to have a better look on issues such as rotordynamic stability, different loads on components such as pressure handling parts, seals and thrust bearing.

Although aerodynamic induced vibration can be evaluated during FLFPFS tests, this should be done carefully as there are no requirements for flow similarity between test and design conditions. A comparison of predicted aerodynamic cross-coupling between specified and test condition may be done for demonstration of relativity. For the same reason, the compressor performance is not evaluated, relative to the guarantee power, as in an ASME PTC 10 Type 1 test. However, hydraulic performance may be compared to the predicted at test conditions which could indicate problems such as abnormal internal recirculation or dirt accumulated inside the compressor.

Full Load / Full Pressure / Full Speed / Full Density

Full density has been used to mean full discharge density or for the average value of the suction and discharge densities the highest of these averages. The first four authors company previously used a full discharge density requirement with good results. The last two authors company uses an average density requirement, as does API 617.

The added requirement to the test gas density improves the evaluation of rotordynamic behavior, since most of the aerodynamic induced forces will be present. Stability can be checked according to the API criterion shown in figure 1.

Although this test is not performed in similitude with the rated site conditions (as Type 1 requires), a scheduled FLFPFSFD test can be modified to result in a test condition that is close to similitude, enabling the qualitative evaluation of aerodynamic issues, such as rotating stall. Such an evaluation is useful if during the PTs, with vibration criteria as described by Ishimoto et al (2012), there were indication of such problems.

Stability measurement

A stability test is the measurement of the compressor logarithmic decrement. It has typically been done using a magnetic bearing to apply a nonsynchronous forced excitation on the rotor. The main objective is to provide a stability map of the compressor, i.e. to know previous to field operation in which regions the compressor can operate safely. This is especially important in upstream projects because there is may be a considerable uncertainty about the gas composition.

Moore, et al (2002) describes how the development of hole pattern seals with shunt holes and swirl brakes at the division wall or balance piston inverts the known stability curve with toothed labyrinths. With the use of these designs, the log dec is actually increasing with the pressure and density. But with toothed labyrinths the log dec should decrease with increasing density.

The hole pattern or honeycomb seals need shunt holes and/or swirl brakes to avoid instability as described by Memmott (1994), Gelin et al (1996), and Camatti et al (2003). In all of these cases shunt holes and not swirl brakes were used to stabilize the compressors. The problems were found at the vendor's facilities on FLFP tests and not on the mechanical tests. There was no measurement of the log decs on those tests.

Pettinato, et al. (2010) proposes that the stability measurements can be carried out during a mechanical running test and an ASME PTC 10 Type 2 test. The first method for acceptability discussed by Pettinato et al. (2010) corrects the stability curve with a bias shift based only on measurements obtained with the MRT. The second method for acceptability discussed by Pettinato includes the measurements at the performance test and in this case there is some cross-coupling. The stability curve will be corrected not only by a bias at 0 cross-coupling, a slope correction of the curve will also be done. In this case we have an evaluation of the model of the annular gas seals. With these corrections we can have a better prediction of the stability on site, but we would still have some uncertainties, for example for high pressure or high density compressors, and the FLFP still remains as the best option to do this kind of evaluation.

Complete Unit Test

The experience from end users shows that problems which initially appear to be insignificant can cause huge costs and delays on the startup of a compression system, especially in off-shore applications.

In addition to the problems already cited in this work, Maretti, et al. (1982) shows a list of potential problems that can be detected by performing the full load test with the main equipment and the auxiliaries:

- Drivers and couplings (interaction vibrations, abnormal axial thrusts);
- Control panels and motor control centers (incorrect cabling, instability of regulation);
- Lube and seal systems (improperly sized components, control problems);
- Main skid (misalignment owing to insufficient stiffness, resonance, inappropriate supports);
- Coolers (marginal sizing, excessive pressure losses, bundle tube failure caused by vibrations);
- Vibrations on loaded gearboxes or electrical motors;
- Logic problems (i.e. auxiliary lube oil pump did not start up on the appropriate time);
- Vibration on auxiliary pumps or on shaft driven pumps;
- Validation of consumption of utilities.

Additionally, the test gives a chance to evaluate, as close as possible to site conditions, the control instrumentation setup and operation, start-up and operating sequences of the system.

ASME PTC 10 Type 1 + Stability measurement + Complete Unit Test

This would be the most complete test, basically reproducing the site conditions and confirming the compressor stability map. In this case the equipment performance is fully evaluated and the acceptance criteria from ASME PTC 10 and API 617 are applied. However there are no vibration acceptance criteria in API 617 for this test, just for a mechanical test.

Furthermore, during the scanning of the predicted curve, aerodynamic phenomena such as rotating stall can be better investigated since the test is in similitude with site conditions.

Procedures for determining the accuracy of the Type 1 test should be developed at the proposal stage to measure if the objectives are met. Gas constituency accuracy, equation of state to be used, instrumentation location and quantity all contribute to the overall accuracy.

TEST SPECIFICATION

The differences between the tests usually called “full load tests” can lead to lots of discussion during the project development. Annex I of Part 2 of the API617 8th Edition (2014) has a description of the various FL FP FS tests, but it gives no acceptance criteria, such as on overall vibration limits and limits on non-synchronous vibration. Subsynchronous vibration levels are the major concern. There is no definition in the standards, and therefore the conditions of the test are subject of agreement between the supplier and the purchaser. The main source of discussion is usually a lack of information in the purchase documentation. It is quite common that these tests are requested on the requisition as “full load / full speed test”, without any additional information.

Most vendors already have standardized their own “full load / full speed test” that can offered based on the test bench structure available. The bid price is calculated on this standard test.

Further into the project, when the test procedure is submitted to the purchaser commentaries, it becomes evident that expectations of vendor and purchaser are different.

Even minor deviations from “standard” test can generate the necessity of unforeseen investment costs by the vendor. Equally, limitations on the test bench can inhibit the investigation which was the main motivation of the test purchase.

To avoid all the delay and cost issues (not to mention the stress to all parties involved), the authors recommend that the purchaser issues, as a part of the bid documentation, a description of the required test agenda, including its objectives and any specific requirements or criteria (such as a lower maximum bearing temperature limit or the acceptable log dec measurement methods). The vendor may then provide a test program that best meets those objectives, their limitations and costs, with any deviation to the intended test being discussed prior to the contract signature.

Regarding costs, the purchaser is in a position to evaluate to which point an extended and complex test campaign is worthwhile, customizing test campaign scope to the risks foreseen to the project.

CASE HISTORIES OF FULL LOAD FULL PRESSURE TESTS

This section will describe some case histories of full load full pressure tests. Included will be some problems detected during full load full pressure tests in which a non-load test would not necessarily indicate the issue and some problems which were detected in the field when a full load full pressure test was not done. Some literature review is also presented, but the main focus is based on the recent machines purchased by the first four author's company.

ROTOR STABILITY

Rotor stability might be the greatest driving concern that leads operators to order full load tests. If there is a stability problem (instability) then it should be found by adequately prepared full load test in the OEM facility. If found during the initial startup at the operator's site then it could lead the compressor to be unable to be operated safely, it could force the manufacturer to go back to the drawing board and review the compressor rotordynamic design, most likely resulting in a delay in the start of operations.

The first four authors' have not faced any such issues in their company's upstream projects. In the literature, it is possible to find cases that describe the identification of rotor instability by means of a full load full pressure test. See the papers by Coletti and Crane (1981), Fulton (1984), Shemeld (1986), Memmott (1990, 1992, 1994, 2004), Gelin et al (1996), Camatti et al (2003), and Tecza et al (2004). The paper by Moore et al (2006) describes an instability that was detected in the field, where there had not been a FLFP shop test. The fact that honeycomb seals need deswirling (in these cases shunt holes) is described in the 1994 paper by Memmott, the 1996 paper by Gelin et al, the 2003 paper by Camatti et al and the 2006 paper by Moore et al.

The published cases of instability are mostly old ones, as over the years stabilizing parts have been developed such as squeeze-film dampers, tilt pad oil-film seals, shunt holes, swirl brakes, and damper seals (honeycomb and hole pattern types). Analytical programs are available to model the compressors with those parts. The stability requirements that were put into API 617 7th Edition (2002) and expanded in API 617 8th Edition (2014) give added assurance that stability problems will be identified at the design stage and not in the field.

There are papers that describe FLFP tests where the compressors were stable with no design changes needed. See the papers by Memmott (1990, 1999, 2004, 2011), Bidaut et al (2009) and Bidaut and Baumann (2010) and the papers listed in the next paragraph and the following two examples.

It was approximately 30 years after FLFP tests were first done before there was a measurement of the log dec on such tests. Papers describing how the log dec with deswirled hole pattern seals increases with increasing pressure and density were written by Moore et al (2002), Moore and Soulas (2003), Gupta et al (2007), Soulas et al (2011), Memmott (2011) and Colby et al (2012).

We describe recent successes in full load full pressure testing in the shop of compressors with hole pattern seals for applications which involve high pressure high density CO2 rich gases and the application of magnetic bearing exciters to measure the log dec vs. load, pressure and density. The compressors were stable with the parts that they were designed to run with.

The paper by Colby et al (2012) describes the magnetic bearing exciter testing in the OEM facility for 3 different compressors, called Tupi I, Tupi II, and Tupi III. The mag bearing exciter testing was with a SDOF system. All three were back-to-back compressors with convergent hole pattern seals at the division wall with shunt holes and swirl brakes at the division wall, swirl brakes at the impeller eyes, squeeze-film dampers and dry gas casing end seals. All three had ASME PTC 10 type 1 tests and were stable. The Tupi I compressor testing had also been described in the papers by Soulas et al (2011), Memmott (2011), and Gupta (2011).

Results are given for the Tupi III compressor. The highest MW used was 33.6 MW for a CO2 hydrocarbon gas blend, and with that MW and an inlet pressure of 250 bara (3,629 psia), and a discharge pressure of 560 bara (8,126 psia), there was a highest discharge gas density of 560 kg/m3 (34.97 lb/ft3), and highest average gas density of 530 kg/m3 (33.06 lb/ft3). A magnetic bearing exciter test was done at that highest density condition. A literature search found that the previous highest average gas density achieved was 480 kg/m^3 (30 lb/ft^3).

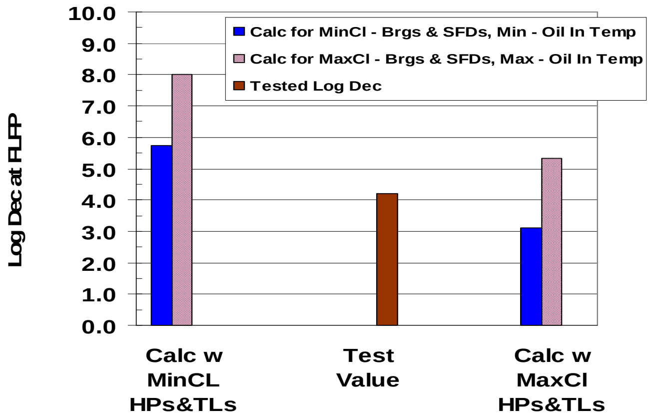

Figure 5 - Calculated vs. Test values of log dec for max tested density for the Tupi III compressor.

The chart in Figure 5 shows the calculated log decs for the highest density condition tested as compared to the measured highest density condition tested as compared to the measured value from magnetic bearing tests for that condition. The compressor easily met the requirement in API 617 that the log dec be greater than 0.1 for the Level II stability analysis. The measured value fits within the range of the predicted values.

Figure 6 show a vibration spectrum from the full-load full-pressure testing for the Tupi III compressors, for the highest average gas density tested. There is no subsynchronous vibration.

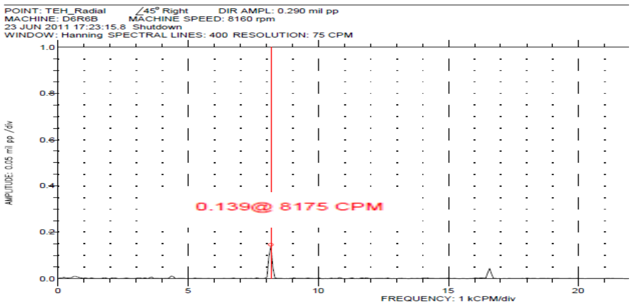

Figure 6 - Spectrum plot for the Tupi III compressor for the max tested density, free of SSV.

Noronha et al (2014) describe the stability analysis and testing of a nine stage back-to-back centrifugal compressor in the OEM facility. This was the high pressure compressor in a train with two compressors compressing CO2 rich gas. The final discharge pressure was 250 bara (3630 psia), but with MW ranging from 26 to 39 (CO2 content from 33% to 83%). It has squeeze-film dampers, convergent hole pattern seals at the division wall with swirl brakes, swirl brakes at the impeller eyes and dry gas casing end seals. Stability measurements were specified by the operator and carried out during the full load / full pressure test and at a low pressure and an intermediate pressure. The log dec measurements were made with a magnetic bearing exciter with a MDOF system. Previous magnetic bearing exciter tests by this OEM were with a SDOF system.

These tests confirmed the trend of increasing log dec with increasing density due to the use of hole pattern seals with shunt holes and swirl brakes at the division wall. See Figure 7 for the variation of the predicted log dec with increasing average gas density. The log decs include the effect of a modal sum of the API anticipated cross-couplings at each impeller as in Memmott (2000). The Noronha et al paper (2014) compares the predicted to the tested values. The testing was done up to a value of 107 kg/m^3 (6.7 lb/ft^3) average gas density. This point was at FLFPFS, but not at the highest average gas density. The last two points are operating points for the compressor. The one on the right has the highest average gas density. There would be no continuity if the plot was versus discharge pressure or load. The horizontal axis of average gas density is the same as in the API experience plot and in Figures 1 and 2.

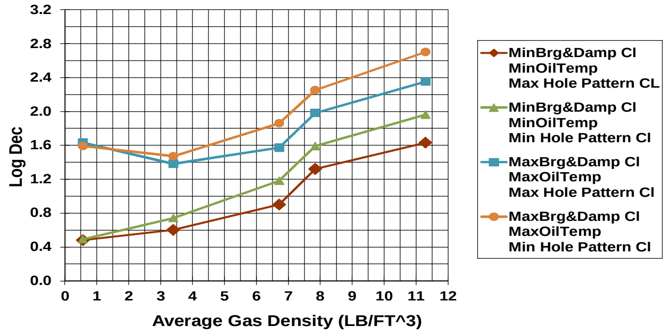

Figure 7 - Predicted log dec vs. Average Gas Density as in Noronha et al (2014)

A spectrum plot from the full load test is shown in Figure 8. The compressor was stable and the purchaser stated that predicted log decs were found to be very conservative.

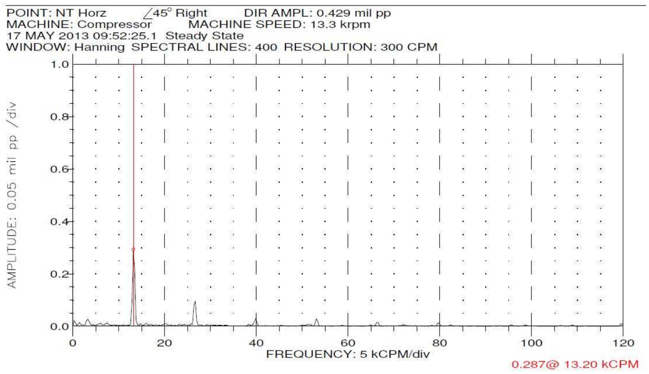

Figure 8 - Spectrum plot from the full load test for the 9 stage high pressure CO2 compressor, free of SSV

AERODYNAMIC ISSUES

Mechanical running tests and the performance tests are typically performed at a very low pressure. Consequently, aerodynamic issues, such as rotating stall, have relatively low energy and may not be identified unless it has a significant impact in the compressor performance or sufficiently sensitive instrumentation is used to watch for such phenomena

Ishimoto et al (2012) reports a case in which subsynchronous vibration was detected during a PTC 10 Type 2 test. This vibration was linked to rotating stall and the OEM decided to pinch the diffusers. The compressor was retested at type 2 conditions, showing very low subsynchronous vibration. Later in the project, the phenomenon was confirmed in a FLFSFD test. The observed vibration values were higher because of the higher density, as predicted, and much easier to identify. The stall inception point was, however, at a higher flow than predicted (most likely due to higher Reynolds number), reinforcing the value of the additional test to assess the impact of similar issues in the compressor operational range.

As described in the cited paper, sometimes it is possible to detect the aerodynamic excitation issues by monitoring subsynchronous vibration during the performance test. However, API 617 does not require the acquisition of vibration data during the performance test. Even if the vibration measurement is required, acceptance criteria is still an issue, since the low density of the test may decrease the vibration to levels where it may be undistinguishable from noise and go unnoticed. Ishimoto et al (2012), proposes the addition of some new requirements to the ASME PTC 10 Type 2 test to deal with this issue.

Sorokes, et al (1994), describes how an ASME PTC 10 Type 2 test can sometimes show no substantial subsynchronous vibration, while, for the same compressor, a test with conditions approaching the ASME PTC 10 Type 1 can demonstrate high levels of vibration within the operational range of the compressor. In the case covered by the Sorokes, et al paper, the subsynchronous vibrations were caused by vaneless diffuser rotating stall or stall in the return bends.

AXIAL LOAD

During the mechanical running test and the performance test the compressor operates at a very low pressure, the axial bearing is operating practically without load.

On a recent project there was a problem with the axial bearing. The mechanical running test and the performance test showed a normal bearing pad temperature; however, when the FLFSFP test was done, the axial bearing temperature increased to values above the specified limit. The bearing design was modified: the material was changed to a copper alloy to improve the thermal conductivity as shown in figure 9.

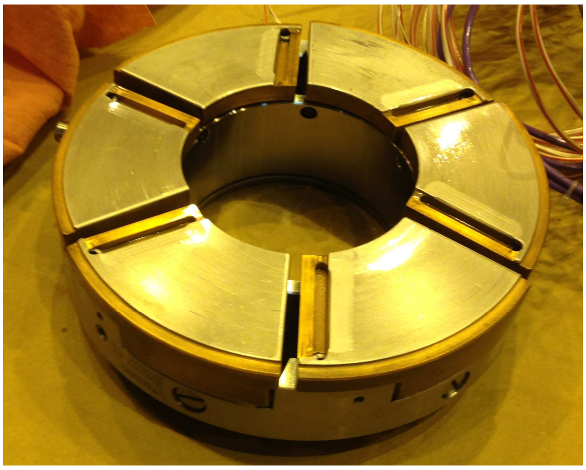

Figure 9 - Bearing with copper alloy.

In another similar case of excessive load on the axial bearing, with high pad temperature in the axial bearings during a FLFSFP test, the thrust balance was modified by small but important changes to the division wall and impeller eyes. The impellers were reworked and new labyrinths and a new division wall were manufactured.

In other cases, when the axial bearing is not loaded, it can actually induce some subsynchronous vibration. The full load test can be used to confirm that this subsynchronous vibration is caused by this condition and eliminate any concern regarding other problems as shown in figures 10 and 11

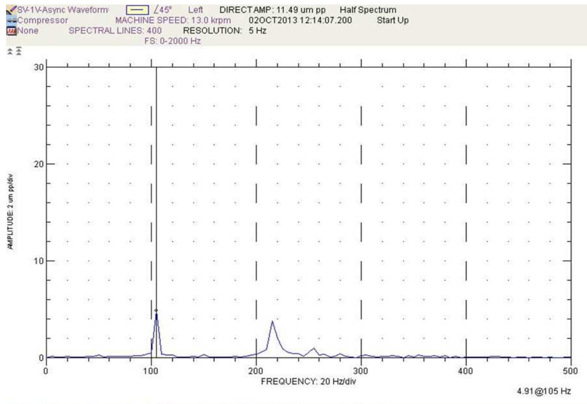

Figure 10 - Spectrum showing a SSV during low load test.

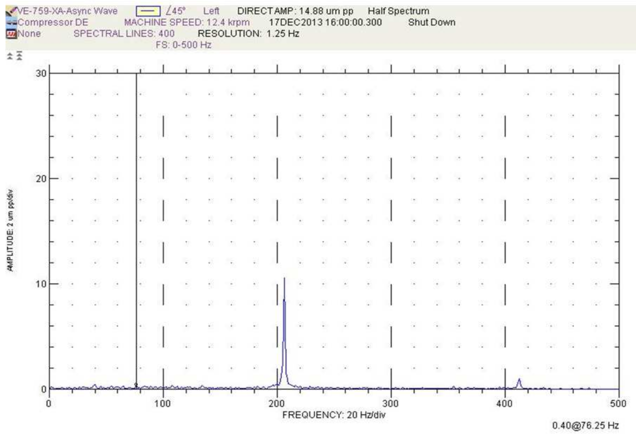

Figure 11 - Spectrum of the same machine during the full load test, no SSV detected.

PRESSURE RESISTANCE

A critical point that cannot be evaluated during the mechanical running test and performance test due to the low pressure is the mechanical integrity of internal components. Those components are not subjected, during reduced pressure tests, to the mechanical loads in which they will operate at site. Even the gas leakage test, being performed either at the maximum sealing pressure or the maximum discharge pressure, does not subject the internal compressor components, other than the casing joints, to operational loads, since the pressure is equalized across the compressor.

DRY GAS SEALS

In a recent case, a dry gas seal failure occurred during an OEM internal FLFSFP test, prior to the client witness testing. The root cause analysis revealed a seal design weakness, where a small leakage on a polymer seal would cause over pressurization of the barrier sleeve. The redesign of the seal included a secondary elastomeric seal, with any leakage being vented to the primary vent.

THERMAL BOW

A hot restart condition, i.e. a restart before the end of thermal transient, can lead to a thermal bow and consequently high vibrations.

The MRT and PT could not detect this problem, as the hot restart condition it is typically not mandatory. Depending on the time to install the unbalance weight, during the URT the thermal transient could cause a rotor bow. However the thermal gradient under vacuum or low pressure helium is quite different from the pressurized conditions.

Baldassarre and Fontana (2010) described mathematically the rotor bow during the hot restart of centrifugal compressors. A simplified criterion to screening the rotors that may be susceptible to the rotor bow was proposed.

The FLFP + CUT will normally foresee a hot restart condition, as normal shutdown and some emergency stop (i.e. push emergency button) followed by a restart are normally asked to validate the control logic.

Figure 12 shows a FSNL test which presented a thermal bow. In Figure 13, showing the synchronous vibration fall under constant speed, the confirmation of such behavior is shown (bow being rectified).

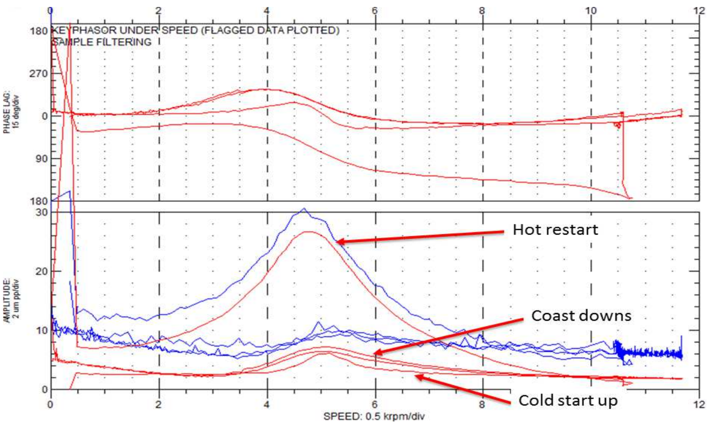

Figure 12 - Bode at ramp up and coast down.

As this test was an open air low load string test and the casing temperature was higher than the expected at operation, doubt existed that the problem would occur in the field. A FLFSFP could give the information if any countermeasures, such as slow roll implementation or interlock the standstill time, may need to be applied (with operational impact).

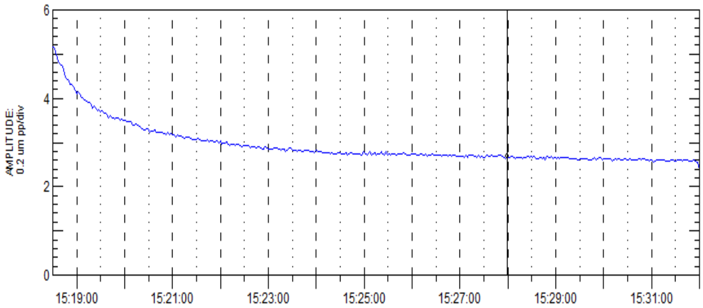

Figure 13 - 1X Vibration fall at constant speed (over time).

DAMPER SEAL EFFECTS ON UNBALANCE RESPONSE

The use of damper seals, such as hole pattern and honeycomb seals with shunt holes and/or swirl brakes, has been a trend in centrifugal compressor design. They greatly increase the system damping, reducing the possibility of having an unstable rotor. However, even if they have a great impact in the rotordynamic behavior, they are not taken into account in the current API 617 standard damped unbalance response analysis (although the API RP 684 states that they can have significant impact). When it is accounted for, in the API 617 level 2 stability analysis, it is part of a search for the system eigenvalues and whether they are stable or not, with no evaluation of resulting rotor mode shape and unbalance sensibility at design operation conditions.

In one of the operator’s projects, the centrifugal compressors, which had only been through the standard test agenda, without any high pressure test, started operation with high vibration at the non-drive end of the high pressure casing. Particular to the vibration behavior of these machines was a clear pattern of increasing vibration with the discharge pressure (Figure 14). Since the issue was only discovered during the unit start up, there were only limited conditions available (both in terms of tools and production availability) to investigate its cause, with the unit being operated at very narrow constraints.

Fortunately, at the same time the units purchased for two other production platforms from the same OEM, and with essentially the same rotordynamic design, were undergoing factory acceptance tests, which included a FLFSFP test. When this full load test was first performed, the issue occurring on the operating machines was clearly observed. After the investigation by the OEM, the vibration was attributed to an overly stiff hole pattern seal. The compressor has a straight-through design with the hole pattern seal at the drive end making it work practically like a third bearing (stiffer than the actual ones), bringing the rotor to an “overhung” condition at the non-drive end and making this (suction) side very susceptible to the residual unbalance.

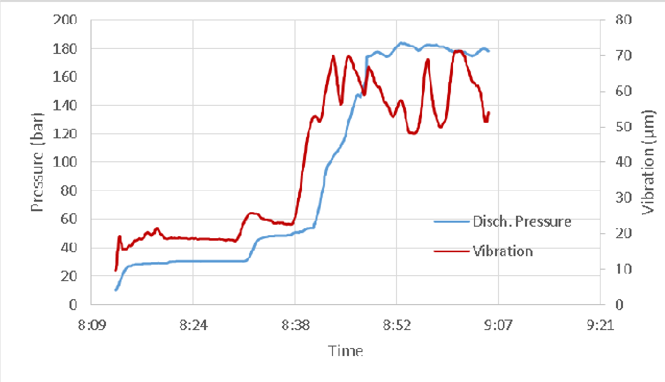

Figure 14 - Compressor behavior after startup.

The hole pattern seal design was then modified to reduce the stiffness by replacing the initial third of the hole pattern seal length by a toothed labyrinth seal (Figure 15). The FLFSFP test was rerun with this modified design, showing vibration at acceptable levels. With the favorable test results, the modified balance piston seal was applied on all nine compressors, including the retrofit of the operational ones.

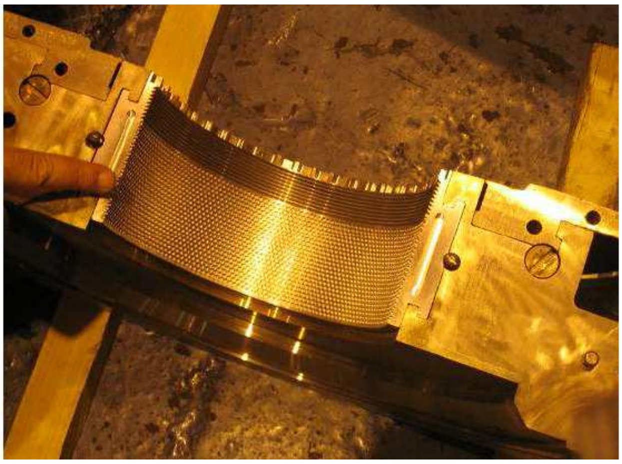

Figure 15 - Seal after first modification.

In the field, the machines with the modified balance piston seal presented mixed behavior. Some units presented overall low vibration, while others showed the same, although less severe, behavior of high vibration tracking the discharge pressure. For years, the issue was managed with special operational procedures and more frequent maintenance interventions for the units most prone to high vibration. After an especially unsuccessful maintenance effort, with one of the bundles, even with clearances, runout and balance measurements according to design tolerances, the unit experienced trip level vibrations upon re-start, so the issue was again revisited.

The operator and the OEM jointly reviewed the hole pattern seal design, evaluating possible changes in the design clearance and taper angle. In the end, the modification, which could be applied at the operator workshop to the existing spares, consisted of simply machining part of the seal length, removing a portion of the hole pattern so it was now in just one third of the original design length, as can be seen at Figure 16.

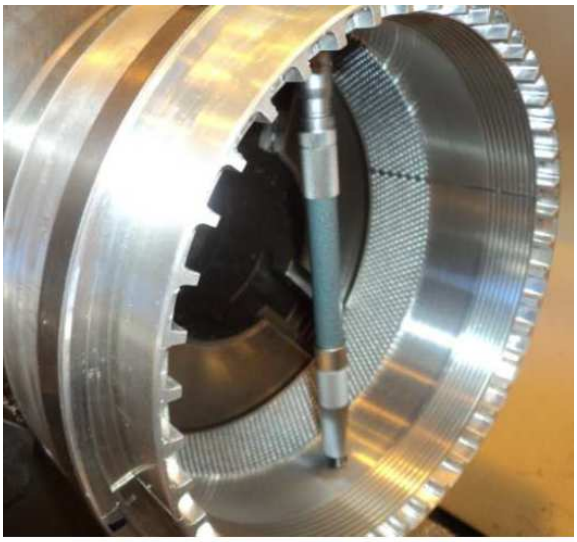

Figure 16 - Seal after second modification.

The problematic bundle was then reassembled with the new version of the balance piston seal and put into operation with impressive results. Greater vibration decrease was observed than after the first hole pattern seal design modification (Figure 17). After six months of reliable and stable operation, the operator has decided to extend this new hole pattern seal modification to the remaining compressors on future workshop maintenances. Noronha et al. (2013) presents with detail this case.

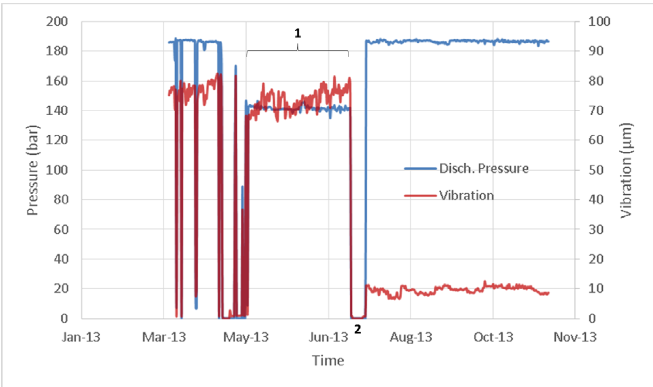

Figure 17 - Non-drive end vibration and discharge pressure relation prior and after the second seal modification. The trend shows the period of time (1) during which the just maintained compressor had to be kept at reduced discharge pressure (with an associated production loss) to avoid vibration trips until the modification was performed (2).

LOW FREQUENCY INSTABILTY DUE TO DAMPER SEAL NON-CONVERGENCE

The paper by Memmott (2012) shows how undamped critical frequency analysis can be used to understand a low frequency instability problem that has been seen a few times in the past in centrifugal compressors with hole pattern or honeycomb seals at the division wall or balance piston. This has been found either in full load full pressure tests or in field operation. The problem has occurred when the seal is built straight or calculated to be divergent (equal or larger clearance on the low-pressure side as compared to the clearance on the high-pressure side). This situation may result in a negative direct stiffness at these seals. The negative stiffness can produce a low frequency mode with unacceptable levels of subsynchronous vibration at that frequency. This low frequency mode is seen to be the first fundamental bending frequency. The solution to the problem is to make the seal convergent (smaller clearance on the low pressure (outlet) side as compared to the clearance on the high pressure (inlet side.)

The undamped analysis in the 2012 paper was done on a compressor discussed in Eldridge and Soulas (2005) and in Memmott (2012), The compressor had been revamped in the field to six impellers from three and there was the addition of a straight hole pattern seal at the balance piston with shunt holes and swirl brakes and swirl brakes at the impeller eyes. It always had squeeze-film dampers and dry gas casing end seals. The discharge pressure was not very high, 83 bara (1200 psia). When the revamp was started up it had a low frequency instability at 6% of running speed. The hole pattern seal was redesigned with a convergent taper at the balance piston and this removed the SSV. See Figures 18 and 19 for before and after. The stability analysis, design process, and implementation of the convergent taper hole pattern seal in the field took less than a week.

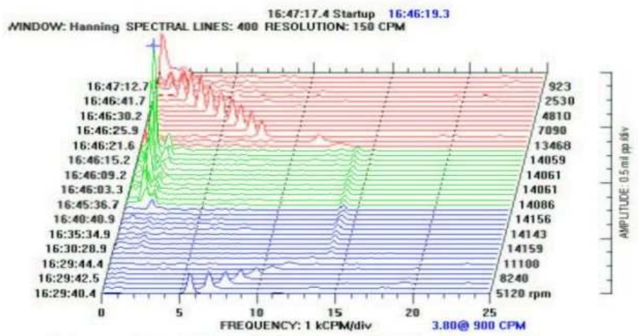

Figure 18 – Low Frequency instability with straight hole pattern seal at the balance piston from Eldridge and Soulas (2005) and Memmott (2012).

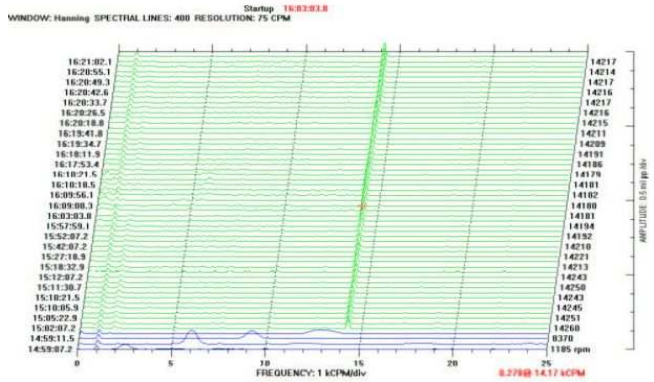

Figure 19 – Stability with convergent hole pattern seal at the balance piston from Eldridge and Soulas (2005) and Memmott (2012).

Other examples of such behavior observed during the testing campaign or field operation and how it was tackled are discussed in Camatti et al (2003), Kocur and Hayles (2004), Tecza et al (2004), and Moore et al (2006).

The papers by Bidaut et al (2009) and Bidaut and Baumann (2010) have extensive discussion on design considerations for hole pattern seals.

The 8th Edition of AI 617 has requirements on the analysis of compressors with damper (hole pattern and honeycomb) seals that should take care of the low frequency instability problem that has been seen with honeycomb or hole pattern seals.

OBJECTIVES, PROCEDURES AND CRITERIA

Colby (2005) states that the objectives of the tests are usually not well defined in the proposal phases of the project. A good description of the test, as part of purchase documentation, is essential to the success of the test. In this section, some important points to be established during proposal phases will be discussed.

Test objectives

This is the main question to be answered by the purchaser when ordering a full load test. What are the foreseen project risks that the purchaser wants to mitigate by testing?

For example, if there is a risk of a phenomenon like rotating stall, the test must thoroughly investigate the low flow region of the performance map. If the concern is stability, because of i.e. a very high density of the gas, then a full density test is a good option, but a stability test is even better. The complete unit test, especially if done at the shipyard, speeds up the commissioning process of the train by anticipating part of the efforts.

Job & Test equipment

One of the first steps is to clearly define the systems and subsystems that will be included in the test scope. In other words, it shall be defined which equipment and subsystems used in the test must be from the job and which may be the OEM's standard test systems.

In general, the purchaser would want to have a test configuration as close as possible to the field configuration. However, this can significantly increase the complexity of the test (and therefore its costs) and may cause some procurement difficulties.

The suggestion here is to carefully evaluate which of the system components can have an impact on the foreseen project risks, and to use this evaluation to list the components that must be included in the scope of the test.

Acceptance Criteria

The parameters to be used as acceptance criteria for the test should be clearly defined. Normally, overall vibration (shaft and casing) and bearings temperatures are included, with purchaser and OEM discussion regarding the acceptance level. The range of proposed level can vary from API 617 limits for mechanical running test up to the operation alarm values. The first four author's experience says that 120% API's mechanical running limit for overall vibration is achievable during the high pressure tests and ensures a reliable machine.

Other vibration criteria such as non-synchronous vibrations limits or limit for vibration during excursions through the first critical speed may also be included depending on the test type and purchaser concerns. Specifically regarding non-synchronous vibration, the first four authors have good experience using an acceptance criterion of 20% of the overall vibration limit, the same as in the mechanical test requirement in API 617.

Regarding bearing temperature, the same criteria used during the mechanical running test may be used for full load tests. When a different oil grade is used for testing, the bearing temperature limit may be adjusted accordingly with the supply temperature, keeping the same supply/bearing temperature increase.

A common criterion, especially for CUTs, is that any non-scheduled alarm will cause the rejection of the machine.

Test Type

As described earlier, there are some different types of tests that can be performed depending on the combination of parameters such as full load, full pressure and full density.

The first point that the purchaser shall define is which combination fits better the project risks in discussion and a criteria for each parameter. The general aim is to make the simplest test that covers the purchaser's concerns.

Below we discuss some points that the purchaser should define in a clear way to avoid later discussion on the test procedure.

Full Pressure

The purchaser shall indicate what should be considered "full pressure". Normally the discharge pressure will have a zero negative tolerance, while the suction pressure could have a different tolerance in order to keep the pressure rise through the machine, to verify the Kirk parameter (Kirk and Donald 1983) and to control the flow through the balance piston. Any variances in pressure may also affect the axial thrust loading, an objective of the test. Units having multiple sections should be reviewed to ensure thrust balance is considered in the test design. For new designs applying damping seals, in which the log dec increases with the delta pressure, one condition with a higher suction pressure might also be of interest.

Full Density

As explained, API 617 does not mention this term. Miranda and Noronha (2007) stated that a good strategy to test stability conditions involves the definition of a density tolerance. Either suction/discharge average density or discharge density may be used as the ruling parameter, depending on the goal of the test.

Stability measurement

The first four author's company started to request this kind of test in recent projects considering the following conditions: log dec calculated on Level II stability analysis < 0.2 or high pressure and high density compressors falling far from the region A in the API plot.

The purchaser shall define operational conditions in which the log dec will be measured as well as the methods to calculate the log dec. Pettinato, et al. (2010) and Noronha, et al (2014) indicate the use of MOBAR (Multiple Output Backward Auto Regression). The PEM (Predicted Error Method) is also considered to be reliable. Some advances using the OMA (operational modal analysis) have been achieved, with the advantage to measure the log dec without an external source of excitation.

Test Operating Conditions

Maretti, et al. (1982) states that the best method of performing a full load test consists in scanning the whole compressor characteristic curve, from high flow to surge, over the whole speed range. This sweep is especially useful if the test objectives include scanning for aerodynamic instabilities (which require similitude between test and site conditions) or probing the rotordynamic behavior sensitivity to changing differential pressure across damping seals.

The purchaser should specify the characteristic curves points and speeds to be included in the test. Requesting the manufacturer to plot performance curves for the test gas may also be considered.

The oil supply temperature and pressure is of particular interest, since it can affect the stability and vibrations behavior of the machine. The operator practice is to vary oil supply temperature and pressure to their operational limits, covering all the combinations between oil operational pressures and temperatures (the same variation is specified by the operator to the mechanical running tests).

The gas temperature is also important and it should be analyzed. There are two opposing views here.

The limitation of gas temperature variation aims to reproduce the thermal effects across the machine. As the clearances are tighter and tighter, and the pressures are getting higher some discrepancy on the casing temperature can lead to substantial changes on the leakages, and consequently at the performance of the compressor. Higher temperatures may also lead to thermal bow and hot restart issues.

On the other hand, inert gas test will always be at a higher temperature, and the decision to control the temperature will force a more expensive test on hydrocarbon gas.

Considerations for Complete Unit Test

When specifying a complete unit test the purchaser should consider that the objective is to obtain a test as close as possible of the startup at site.

In view of that, commissioning activities should be requested. Those activities are alignment, lube oil flushing, wiring check, and others. Reports on these activities should be given to the purchaser's inspector before the test.

During the test, checks on alarms, control, change overs between pumps/filters/coolers should be performed.

Normally seen as a huge effort, since it may imply in commissioning the unit twice (once in the factory and again in the site). Much of the commissioning for the test will be final, if this test is done at an early stage at the final field construction site.

RETURN OF INVESTMENT

The decision of requisitioning full load tests should be taken after a careful risk/reward analysis. The purchaser engineering teams should ask themselves questions like: What are the risks foreseen for the specific project? What are the efforts, both time and cost-wise, that the purchaser is willing to undertake to early detect potential problems? What are the tools and logistics available in the site to investigate and solve an eventual issue? Which would be the operational consequences of such event?

Cost

The cost of a FLFS + CUT can vary significantly depending on the test scope, the availability of an adequate test bench at the OEM factory, the logistics involved and others.

In general, the main compressor vendors already possess an infrastructure that can be used to this kind of test, although some modifications may be needed to meet specific requirements.

A realistic estimative of the test cost is to consider an increase of 8 to 15% in the value of the compression train, when there is not the necessity of relevant investments (i.e. new test bench) to implement the test agenda. Time-wise, one to two months increase in delivery time should be expected per machine tested.

Operating conditions

Normally, the acquisition of a high pressure test is more justifiable when the compressor will operate in a pressure/density condition that is far from the mechanical running and performance test conditions.

Proven experience vs. New designs

When a new compressor is on or outside the boundary of any reference diagram of the supplier, whatever is the characteristic, a high pressure test is recommended.

The more experienced is the vendor and the end user in that specific application, the less the investment in a high pressure test will be needed.

Quantity of units

As the number of identical units rises, the potential losses multiply by the same factor. As normally just one test per compressor type is necessary to validate the design, the cost is "diluted" in the various identical machines.

Site location and impacts on production

For upstream operations, downtime in compression modules is equal to loss of oil and gas production and may even lead to production platform shut-ins (due to loss of gas lift or local gas flaring restrictions). This may mean a loss of revenue even bigger than the cost of the compressor itself. This is especially true for offshore installations, where any investigation and implementation of solutions requires more time and effort. In such cases, high pressure tests are good option to anticipating issues.

Final decision

The final decision shall take in account the risks, the project schedule, and all others factors quoted above.

A calculation of costs of tests and losses generated by one of the problems listed on this paper is unnecessary, because the cost of losing of thousands of oil barrels per day in production (both avoided and occurred) is a few orders of magnitude above the costs of such tests.

The appropriate analogy is to compare the high pressure tests with buying insurance. In many cases there will be an extra cost with no tangible advantage (no problems found), and everyone will be happy with a successful test. However, when a problem is found, all parties will be relieved of finding the problem inside the OEM facilities.

As a final number, from the operator's last 24 distinct compressor designs (purchased in 9 projects for a total of 21 production platforms), in 7 designs important issues were found either during a high pressure test or only detected on the site because of such test was not performed.

CONCLUSION

The decision of ordering or not a full load full pressure test depends strongly on the project – risks, costs, schedule, uniqueness, etc. Nevertheless the recent project history, the culture of the company (cut costs or play on the safe side) or even tradition can play a role on this decision.

This paper shows some problems that cannot be detected by the conventional API performance, mechanical and rotor response verification tests. If this is all that is done then there could be a possible adverse impact in an offshore production system.

There are many different tests generally known as full load (high pressure tests), and therefore the definition of a clear objective for the test is required for the determination of which test will be carried out.

The issue of a test requisites or guidelines by the purchaser during the bidding phase is a practical solution to guarantee that the objectives of the test campaign will be fulfilled.

The oil production business is intrinsically risky and there are many uncertainties in the development of an oil field. The returns, however, are extraordinary. In order to maximize such returns and reduce the overall project risk, an extensive test campaign for the critical equipment should be carefully considered.

NOMENCLATURE

CUT = Complete Unit Test
EHS = Environment, Health, and Safety
FD = Full Density
FL = Full Load
FP = Full Pressure
FS = Full Speed
k = Isentropic coefficient
log dec = Logarithmic decrement
MRT = Mechanical Running Test
MOBAR = Multiple Output Backward Auto Regression
NL = No Load
OEM = Original Equipment Manufacturer
OMA = Operational Modal Analysis
PEM = Predicted Error Method
PT = Performance Test (ASME PTC 10)
SSV = Subsynchronous Vibration
ST = Stability Test
Type 1 = ASME PTC 10 Type 1
Type 2 = ASME PTC 10 Type 2
URT = Unbalance Response Test

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